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CABLE-TOWED OCEANOGRAPHIC INSTRUMENTATION SYSTEM DETAILED DESIGN

Prepared under Contract Nonr 3201(00)
Sponsored by the
Office of Naval Research

SYSTEMS ENGINEERING DIVISION
PNEUMODYNAMICS CORPORATION
BETHESDA, MARYLAND

Technical Report 6691

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February 1964

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SUMMARY

Salient features are described of a cable-towed oceanographic instrumentation system designed for the taking of sound-speed measurements simultaneously and continuously at six depths, to a maximum depth of 5,000 feet from a vessel proceeding at 7¼ knots. A novel housing for the sound velocimeter, designed to minimize sound-speed measurement error due to towing speed, is described. The housing is used to prevent rotation of the cable. The equations for prediction of cable rotation are derived and measurement of the rotational characteristics of a double-armored cable are reported. Description is also given of the development of the electrical signal-transmission system, which incorporates a novel low-distortion amplifier.

The report also contains brief commentaries on installation considerations, operational considerations, and certain aspects of the design and a list of drawings and specifications required for fabrication.

INTRODUCTION

GENERAL

A cable-towed oceanographic-instrumentation system (CTOIS) that provides means for distribution of a number of sensors over the depth-span from the surface to 5,000 feet (at a towing speed of 7 knots) has been developed and designed. The electronic design and certain aspects of the mechanical design are directed toward satisfaction of the particular requirements of investigators at the Hudson Laboratories of Columbia University. The system otherwise remains suitable for towing any compatible instrumentation.

This report contains an account of the salient features of the final design and of certain basic engineering considerations. Since the major developmental and design considerations have been reported[*1,2,3,4] and a system specification[5] issued, reference will be made thereto as required for background information.

SYSTEM-DESCRIPTION

The system, as designed, conforms closely the one resulting from the preliminary design[2] and the subsequent specifications[5]. It consists of a faired, 3/4-inch-diameter towing cable maintained at depth by a depressor[3,4]. The

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* Superior Numbers refer to the references on page 41

cable is made up of finite faired lengths coupled by combination strength and electrical connectors. Means are provided at each connector assembly for attachment of a trailing housing which contains an American Car and Foundry Corporation Model TR4 Sound Velocimeter and a Borg-Warner Corporation Model 8150 "Vibrotron" pressure transducer.

The faired towing line is directed over a sheave over-hanging the stern of the host vessel and thence through a Western Gear Corporation Linear Cable Engine[6] to a storage unit.

Each pair of sound-velocimeters and pressure transducers operate at well separated center frequencies. The signals from each pair are transmitted over the same wire (four signals per pair) and separated on shipboard by suitable filtering. The electrical core of the towing cable is designed to accommodate six units. Twelve signals may thus be continuously monitored.

Expected depth-speed characteristics are shown on Figure 1. Module station locations specified by Hudson Laboratories are shown on Figure 2.

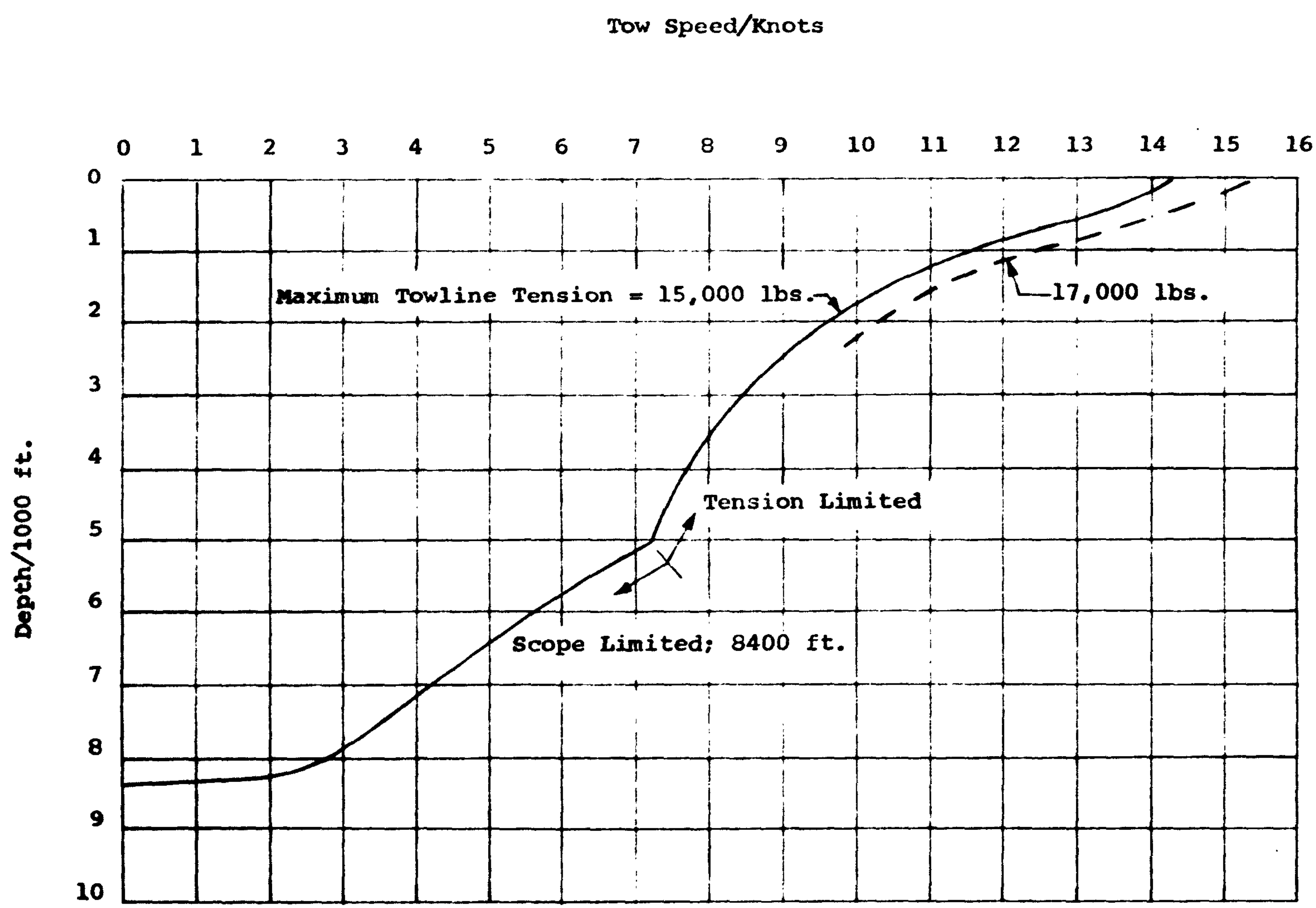

FIGURE 1

Maximum Depth-Speed Envelope For The Cable-Towed Oceanographic Instrumentation System

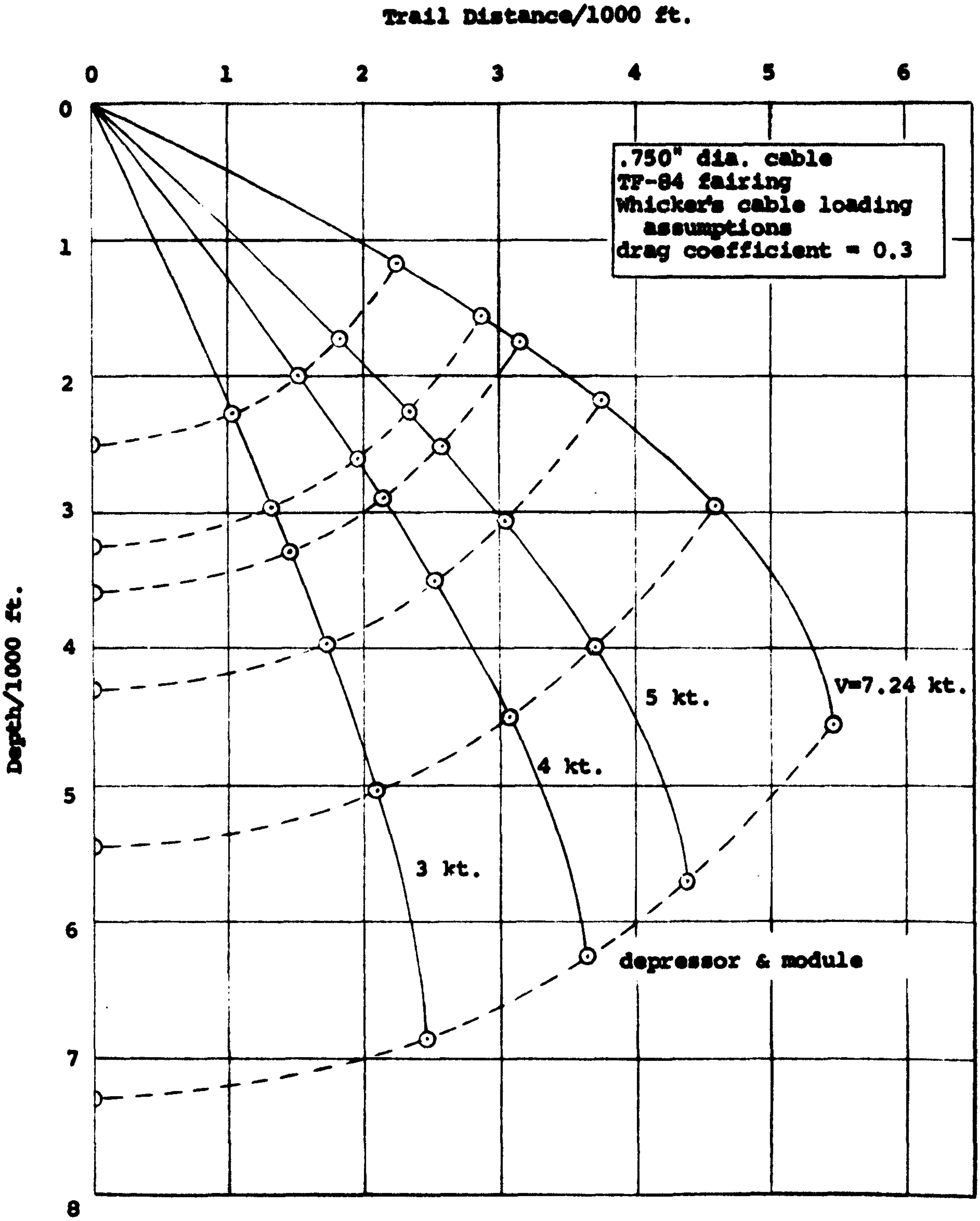

FIGURE 2

Cable-Towed Oceanographic Instrumentation System
Cable Profile At Various Tow Speeds
(module spacing selected to achieve indicated depth at 4 knot tow speed)

ADDITIONAL DESIGN CONSIDERATIONS

GENERAL

In the execution of the mechanical design it was found that certain minor deviations from the specification were desirable for simplification and cost reduction. A major change was introduced in the electronic design, however, to permit continuous, rather than sequential, monitoring of the towed instrumentation. These are reported in this section.

MECHANICAL COMPONENTS

Towed Instrument Module - Specification[8] MS110-1.2 Method of Attachment to the Cable.

The objective that necessitated specification of a towyoke (Section 3.2.1.3 of the referenced specification) was attained by closely coupling the instrument housing to the cable at a point above the connector assembly, thus placing the sound Velocimeter in a relatively undisturbed flow as illustrated on Figure 3. The signals from the module are transmitted to the main cabling via a "jumper-cable."

It was originally planned that the module would be permitted to freely swivel relative to the tow cable and that a length of the jumper cable would be provided sufficient to permit it to wrap about the cable connector as the main cable rotated to an equilibrium position.

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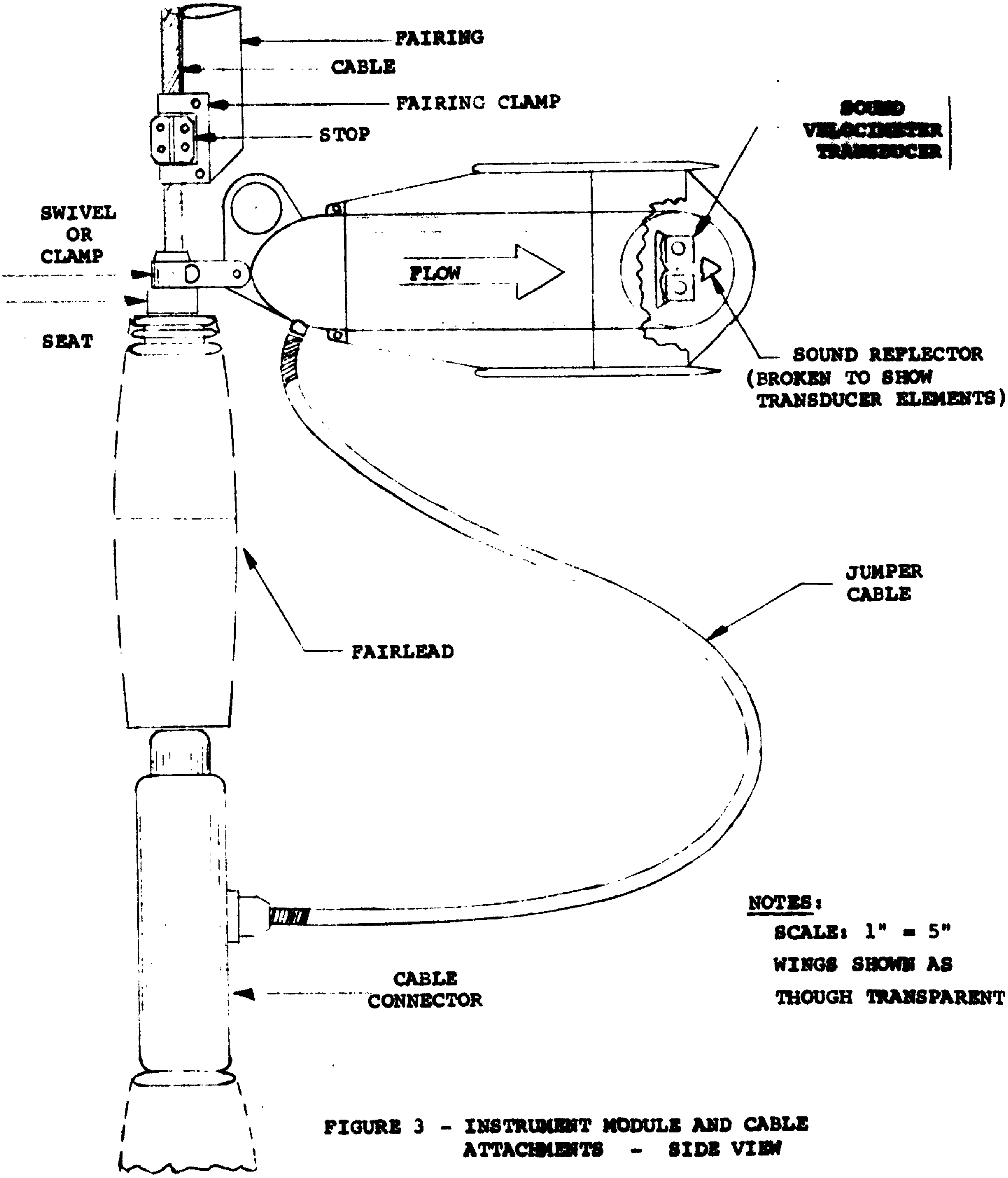
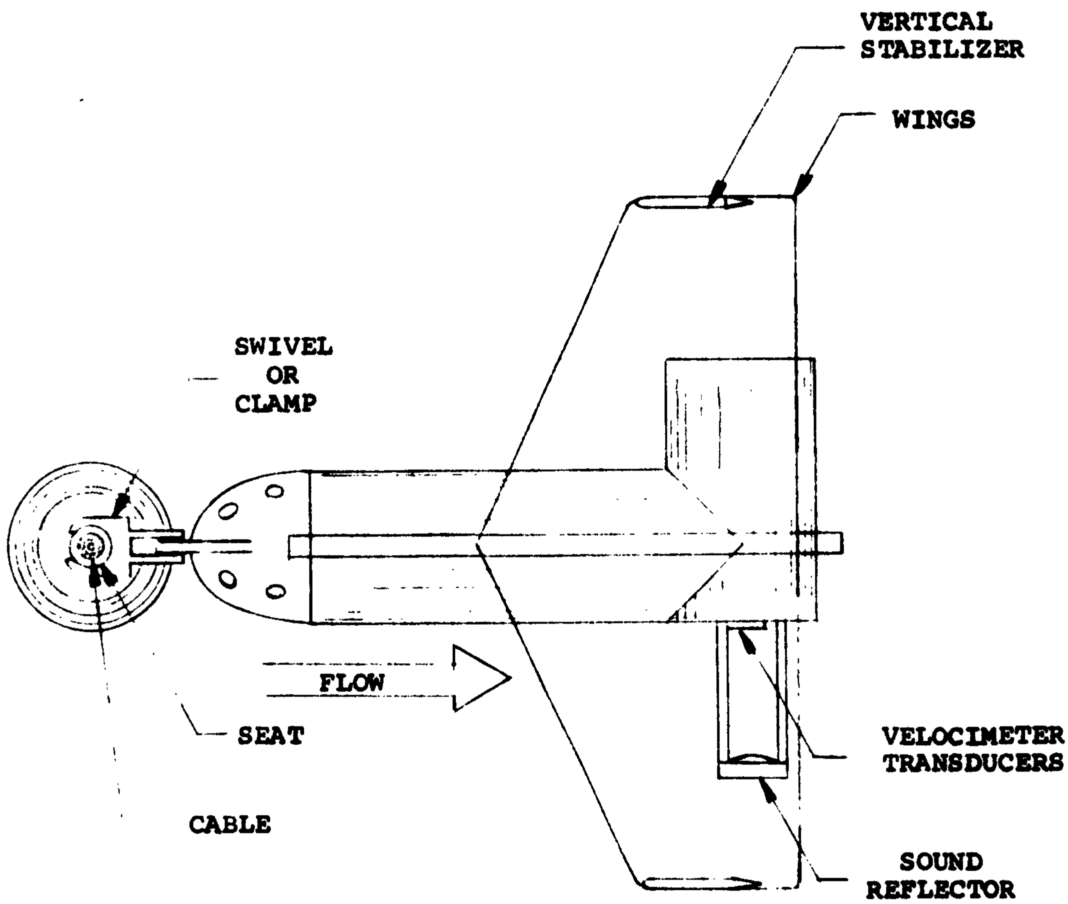


NOTES:

SCALE: 1" = 5"

WINGS SHOWN AS THOUGH TRANSPARENT

FIGURE 3 - INSTRUMENT MODULE AND CABLE ATTACHMENTS - SIDE VIEW



NOTES:

SCALE: 1" = 5"

WINGS SHOWN AS THOUGH TRANSPARENT

FIGURE 3 - INSTRUMENT MODULE AND CABLE ATTACHMENTS - OVERHEAD VIEW

Recently completed work on the rotation of long cables has provided indication that the rotation is too large for convenient use of this technique and that the hydrodynamic force on the module is sufficient to prevent more than a few degrees of rotation at each station. The modules may thus be rigidly coupled to the cable in all degrees of freedom other than in pitch*.

The statement above is based on computed values of the side-slip angle of the module required to null the torque at each station. The torque imposed on the module was predicted from measured values of the parameters required to "characterize" the cable, as explained in Appendix II. The form of these parameters and the relations thereof to values of the rotation and torque in the cable are given in Appendix I. An analysis of the rigid body motion of the module as a result of cable rotation is given in Appendix III.

The values of the cable parameters were also used to predict values of the rotation of the torsionally - non-constrained cable. The result is given in Figure 4. For purpose of this computation, the tension in the cable was assumed to vary linearly with length. The degree of the approximation involved is illustrated on Figure 5.

Expected values of ψ, the side-slip angle; θ, the cable rotation; and L_M, the torque applied to the cable by the module, are given in Table 1 for towing speeds of 3 and 7½ knots.

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* Except the module at station #6, as is explained later.

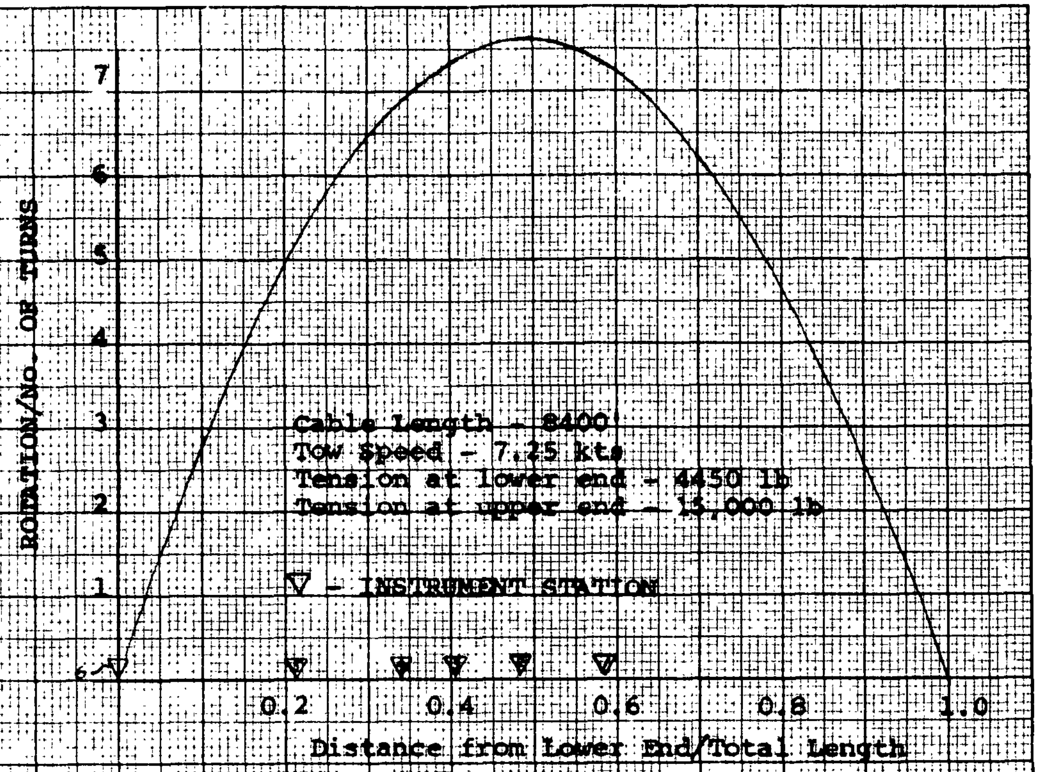

FIGURE 4 - DISTRIBUTION OF ROTATION ALONG CABLE

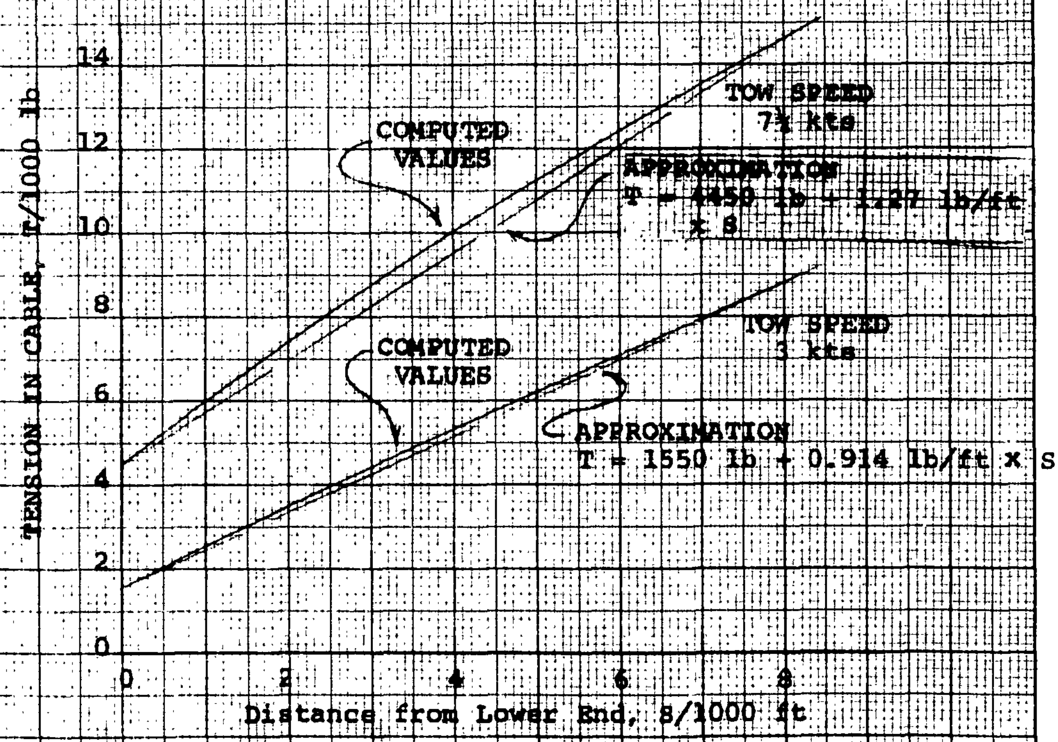

FIGURE 5 - VARIATION OF TENSION ALONG CABLE

TABLE 1

PREDICTED VALUES OF THE SIDE-SLIP ANGLE, ψ, CABLE ROTATION, θ, AND THE TORQUE IMPOSED ON THE CABLE BY THE MODULE, L_M.

STATION	DISTANCE FROM LOWER END OF CABLE/FT.	SPEED - 3 Kts.				SPEED - 7¼ Kts.			
		φ/deg	ψ/deg	θdeg	L_M/lb-ft	φ/deg	ψ/deg	θ/deg	L_M/lb-ft
6	0	87	----	0	3.34*	83.8	----	0	7.9*
5	1800	73	4.62	4.85	1.74	47.6	1.23	1.66	2.7
4	2850	70	2.82	3.0	1.02	38.9	1.04	1.64	1.6
3	3400	69	1.64	2.37	0.78	35.8	0.28	0.48	0.4
2	4050	68	2.78	3.0	0.96	33.0	0.77	1.41	1.0
1	4750	67	6.58	7.15	2.16	30.7	2.02	3.94	2.4
0	7500	--	----	0	10.00**	25.6	----	0	16.0**

* Provided by the Depressor
** Provided by the Towing Vessel

These were computed by equating the torque exerted on the cable by the module (Equation [12], Appendix III) to the excess of internal torque in the cable (Equation [13]*, Appendix I) and solving for the equilibrium angle of rotation. This computation was made on the assumption that the rotation angle is zero at the station immediately below and above the one in question. The value of ℓ in Equations [14.1] and [14.2] (Appendix I) corresponds to the length of cable between the stations immediately above and below the station in question and the value of S is just the ratio of the length of cable between the preceding station and the one in question to that length. The value of G is given very nearly by the ratio of the parameters $\bar{C}$ and $\bar{r}^2$ for this case, being about 39.7 x 10^4 pounds. Use of the approximation $\mathrm{Log}_e\left(\frac{G+1}{G}\right) = 1/G$ is thus justified.

With respect to the module, the value of Y_ψ is estimated to be 3.14 (lb/kt^2) V^2, V being the speed of tow, and of ℓ_ψ , approximately one foot. The value of W is about 18 pounds and of ℓ_W, about one foot. The value of α_0 is estimated to be less than one degree at a speed of 7½ knots and about 7 degrees at a speed of 3 knots. Computed values of the angle of inclination of the cable, φ, are also given in Table 1.

It should be noted that inasmuch as the side force on the cable from the depressor and ship are very small, due to the large moment arms involved, the total side force acting

* With Equation [14.1] for ΔL_0 and Equation [14.2] for ΔL_θ.

on the cable is just the sum of the side forces exerted on the cable by the modules. This is only about 13½ pounds at the 7½-knot speed and 8½ pounds at the three knot speed. Hence, we may expect no discernable lateral displacement of the system due to the nulling of the cable rotation.

As it will not be possible to align the modules precisely with the cable, an estimate of the effect of an initial misalignment on module orientation is required. Computation was therefore made of the shift from the null positions, given in Table 1, due to an initial misalignment. The results are given in Table 2 for unit initial misalignment and for a 30 degree initial misalignment. The value 30 degrees was selected as an upper limit on human error in aligning the module with the cable. These results are based on the assumption that the cable is fixed in rotation at the station immediately below and above the one in question and so are slightly conservative.

It is now apparent that the bottom-most module (station #6) cannot be fixed in rotation as the torsional stiffness of the cable between it and the depressor will prevent relief of an initial misalignment. The bottom-most module is therefore attached so that it may swivel freely on the cable.

DEVELOPMENT

In execution of the design of the module it was found not possible to simultaneously satisfy the requirements that the module be neutrally buoyant and compact and capable

TABLE 2

INCREMENT OF ROTATION ANGLE, $\Delta\theta_M$, AND SIDE-SLIP ANGLE, $\Delta\psi$, PER UNIT INITIAL MISALIGNMENT, θ_{M_o}

SPEED	7¼ Kts.				3 Kts.			
STATION	$\frac{\Delta\theta_M}{\theta_{M_o}} \times 10^2$	$\Delta\theta_M^*/\mathrm{deg}$	$\frac{\Delta\psi}{\theta_{M_o}} \times 10^2$	$\Delta\psi^*/\mathrm{deg}$	$\frac{\Delta\theta_M}{\theta_{M_o}} \times 10^2$	$\Delta\theta_M^*/\mathrm{deg}$	$\frac{\Delta\psi}{\theta_{M_o}} \times 10^2$	$\Delta\psi^*/\mathrm{deg}$
6	100.	30.	99.00	29.7	100.	30.	99.9	29.97
5	0.53	0.16	0.39	0.12	2.42	0.725	2.31	0.69
4	1.64	0.49	1.03	0.31	4.80	1.44	4.50	1.35
3	2.33	0.70	1.36	0.41	6.06	1.82	5.65	1.70
2	2.44	0.74	1.33	0.40	5.53	1.66	5.12	1.54
1	1.72	0.52	0.88	0.26	2.76	0.83	2.54	0.76

* Values given are for $\theta_{M_o} = 30°$.

of withstanding the maximum possible depth of about 8,000 feet (Section 3.2.1.1, Specification MS110-1.2). The decision was made to sacrifice buoyancy for compactness.

The design shown on Figure 6 was accordingly evolved. The proper streaming angle is provided by the bi-plane type lifting surfaces and directional stability by the vertical surfaces joining the tips thereof. The box-type configuration thereby formed serves to duct the flow across the head of the sound velocimeter on a track nearly perpendicular to an axis joining the active and passive elements thereof. Due to the "channeling" effect of the wing structure, the flow will maintain the desired orientation for all angles of inclination of the module less than the stalling angle of the wings.

A high-impact polystyrene plastic was specified for the wing material as it is nearly neutrally buoyant in sea water.

The tow yoke consists of a crescent-shaped piece attached to the nose of the module, as illustrated in Figure 3. The opening in the crescent-shaped piece is designed to pass the cable. The yoke is then dropped onto a "seat" of diameter greater than that of the cable, so that the sides of the crescent embrace the seat. The yoke contains a spring-retained detent that engages one of 24 evenly spaced slots in the seat, thus allowing adjustment of the alignment in increments of 15 degrees. Since the initial misalignment can

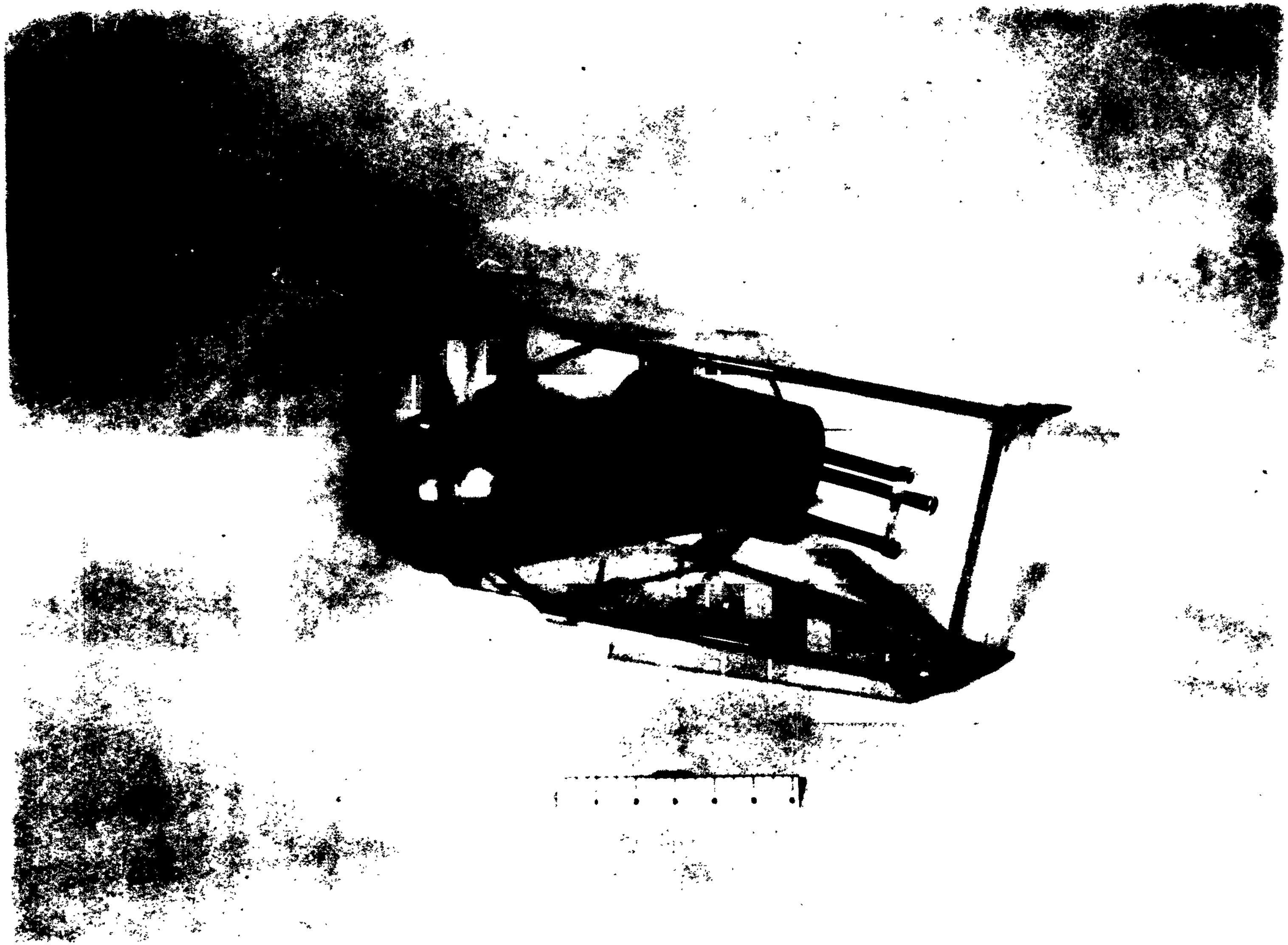

Figure 6 Model of Towed Instrument Showing Location of Sound Velocimeter

be as great as ± 30 degrees without harm, this range of adjustment is considered adequate. The detent is omitted in the yoke for station number 6. The module there is thus free to swivel and align with the flow.

A full scale model, properly ballasted, was towed at speeds up to 7 knots at the David Taylor Model Basin. The model was towed from the mid-length of a cable of prototype diameter and was attached thereto at the nose so as to swivel freely in yaw and pitch. It was observed to be completely stalled at speeds below 1½ knots, but to tow well at speeds greater than two knots. Visualization of the flow across the velocimeter heads was accomplished by affixing tufts of yarn to fine wires stretched across the two front velocimeter posts. At speeds greater than two knots the flow was generally parallel to the wing surfaces and reasonably steady. It is therefore expected that the spatial resolution of the flow and general performance of the velocimeter, as determined by the flow across the heads, will be very satisfactory.

COMMENTS

The side-slip angle of the module is generally acceptable. The side-slip at station number one is considered marginally high at the three knot speed. It can be materially reduced by the simple expedient of mounting a small vertical stabilizer to the upper side of the wing structure. A six-inch square should suffice.

CABLE-CONNECTOR ASSEMBLY AND STERN SHEAVE

Section 3.2.1.4 of Specification MS110-1.1.1.2[5] required spherical swivels at the ends of the connector to prevent bending the cable sharply at the point of entry thereto. A less expensive method for accomplishing this function was subsequently devised and is provided in lieu of the spherically-swiveled ends.

A transition piece consisting of a series of steel discs imbedded in a poly-estane matrix is provided. This piece, designated a "fair lead", is placed adjacent to the connector as shown on Figure 3. The radii of the steel discs are graduated such that when the fair lead is bent against the sheave groove, the center-line of the discs provides a relatively smooth curve with minimum radius of about 20 inches and end-points tangent to the connector-axis and the cable-axis as they lay in the sheave groove. The inner surfaces of the discs are bedded in a poly-estane tube. The poly-estane extends to approximately one-half the disc radius to provide stability and yet allow sufficient flexibility to conform to the sheave-groove radius.

The 3,000 pound-inch torque requirement of Section 3.2.1.4.2 was changed to 1,000 pound-inches as the maximum torque in the cable is estimated to be only 200 pound-inches.

Section 3.3.1.2 of the subject specifications was deleted entirely as entry into the traction unit is now provided by the fair lead.

ELECTRONICS

General

The electronic system orignally proposed by SED was one in which FM subcarrier telemetry data from each of several modules would be sampled. The tow cable would carry power for the electrical equipment in the modules, a shielded data circuit and a set of binary selection circuits. The selected module alone would feed data signals up the cable. A sampling system was considered feasible after consideration of the required accuracy of .01% for sound velocity, 3% for depth, and the expected rate of change of sound velocity with depth. Because only one module would be transmitting data at a given instant, only one set of frequency measuring equipment would be required at the top of the cable. Linearity requirements on cable drive amplifiers would be non-stringent.

After the specification of the electronic instrumentation to measure and record data to accuracies and resolutions compatible with the stated accuracies of the sensors, Hudson Laboratories requested that the instrumentation be changed. The velocimeter data was to be considered as accurate to 1 part in 500,000. A change in the frequency measurement system was proposed which would permit resolution of ± 1 part in 300,000, while retaining a sampled data system. Hudson Laboratories however, requested that continuous sensor data be made available on shipboard. As a result of these requests and further discussion it was decided that:

1. Hudson Laboratories would supply all shipboard electronic equipment except the module electronic power supply;
2. FM data signals from all 12 sensors, (6 velocimeters, 6 Vibrotrons) would be available continuously on shipboard;
3. The tow cable core would consist of 3 coaxes and 3 power leads;
4. The FM data signals from two adjacent modules would be carried on one coax; and
5. Non-standard velocimeters and Vibrotrons would be procured with operating frequencies such that two of each could feed their shared coax with non-over-lapping frequency bands.

Balanced Circuit

Concerning Item 3 above, it was determined that there could and would be interference on the data coaxes since the shield was actually part of the signal circuit. Therefore, balanced data circuits in the form of shielded pairs have been provided in the final design.

Frequency Selection

Concerning Items 4 and 5, the choice of frequencies was made to obtain the least interference from inter-modulation products of those subcarriers sharing the same signal circuit. The one subcarrier frequency accepted was

that of the standard TR-4 sound velocimeter. The frequencies of the Vibrotron pressure transducer and of the other velocimeter and Vibrotron (in the adjacent module) were chosen so that in the velocimeter channels there would be no harmonics nor sum or difference frequencies from any of the subcarriers. At the same time the number of interfering signals in the pressure channels was kept as small as possible. In following this course it was necessary to use two non-standard Vibrotron frequencies as well as one non-standard velocimeter. Before settling on the frequency assignment a reputable filter manufacturer was querried as to the feasibility of the band pass filters required for the velocimeter channels (these being closely spaced). While the feasibility of attaining an accuracy of 1 part in 500,000 for the velocimeter was questioned, the transmission system and frequency assignment have been made so as to degrade as little as possible whatever performance actually obtains.

Line Drive Amplifier

In addition to frequency selection, considerable effort was given to producing low distortion mixing and line drive amplifiers. An initial survey of commercially available telemetry amplifiers disclosed several which could mix signals with the permissible distortion level of 0.1%, but these could not drive a high capacitance, low resistance load.

The amplifier which has been used is an adaptation of a circuit described by Faran and Falks.[7]

It has feedback to the summing input as well as to several points within the amplifier. Unlike many Class B transistor amplifiers, it has a high impedance drive for the Class B stages. The constant current drive which this affords predistorts the Class B input and leads to a quite linear output. Bias adjustment of the output stage was used to further reduce crossover distortion.

Special attention was given to selection of the transformers. The amplifier was to work with a balanced system so it was necessary to have balanced input and output transformers. The output transformer provides an impedance step-up to the line so as to operate at higher signal level.

A further impedance step-up is made in the input transformer to make up for line resistive loss. The frequency range used as a selection criterion was 3kc/s to 25 kc/s. Transformer-produced distortion was required to be less than amplifier distortion. No small toroidal transformers working at the required frequency, power and impedance levels and with such low distortion were found in commercial listings. We therefore wound our own on molybdenum permalloy powder cores. Distortion was found to be less than 0.01% using Hewlett-Packard 302A wave analyzer which has a full scale range of 0.1%, and using a filtered audio oscillator. Smaller cores might have been used but the overall space saving would not have been important.

An ingenious method of oscilloscope distortion checking was devised. The Lissajous method in which the output is displayed against the input is quite familiar. It is useless at the distortion levels encountered in testing this amplifier. An oscilloscope display consisting of input on one axis and difference of input and output on the other proved quite sensitive since considerable gain could be used in the oscilloscope. The difference is taken in a resistive network external to the amplifier.

System Power Supply

An AC power distribution has been used to provide isolation between the modules. Four hundred cycles is used to allow small power transformers and filters. Each module power transformer is connected in parallel across the line. Separate rectifiers and regulators for line amplifier and instruments provide some isolation between measurement and power circuits. The regulators are simple transistorized series regulators with pre-regulators for amplifier stage and with temperature compensated zener reference diodes. Were this job to be done over with emphasis on low power consumption, we would try a variable-duty-cycle circuit with silicon controlled rectifiers. As it is now, fully half of the power consumed is lost in regulation. Although, the regulator supplies a reasonably constant load it must compensate for voltage drop in the supply cable. With six operating modules, this drop could be

20 volts from top to bottom of the 8,000-foot tow cable. The 400 cycle source is a Behlman-Invar vacuum tube oscillator, power amplifier. With power to each module of 5 to 10 watts the line loss would be 6 to 12 watts. A 100 watt supply is ample. Harmonic distortion into a resistive load up to rated power is less than 1%. Unfortunately, the switching transients of the diode rectifiers affect the cable power circuit, which has appreciable resistance. Resistors in series with the diodes limit the transient current. Laboratory checks have disclosed low coupling to the data circuit - on the order of -60 db.

Circuit Construction

Those portions of circuits which must dissipate power such as power transformers and power transistors are mounted on metal plates. Steel is used in preference to aluminum to provide magnetic shielding. There is a plate between the power transformer and the output transformer, and a second between the power transformer and the input transformer - with the input as far as possible from the power transformer. Input and output are both toroids with inherently small coupling. The power transformer core is oriented to give least coupling to the toroids. Other portions of the amplifier and regulator circuits are built on single-sided, glass epoxy, printed circuit boards. The circuits have been vacuum potted in GE-LTV-602 clear silicone rubber. The outer surface has been coated with GE-RTV-60, a red silicone rubber, for protection against abrasion.

During the course of circuit check-out and after potting, operation was checked thru the temperature range of $0^\circ C$ to $50^\circ C$.

Setting Voltage Levels

The line driving amplifier yields a signal with least distortion when the operating level is as high as possible short of encroaching on definite non-linear operation. For a single sinusoidal signal the output maximum is about 8v rms into a shielded-pair cable 1,000-feet long, terminated in 100 ohms. This corresponds to about 22v peak to peak. For a composite signal of 2 or 4 sinusoids this voltage maximum corresponds to about 4v rms.

Access to level adjustment potentiometers -- The amplifier is completely potted. It will be necessary to dig in the potting to gain access to the adjusting screws. The location is given on Drawing SK-1513. The potentiometers are sealed units, but it is suggested that the holes made for adjustment be resealed as noted on the drawing.

Level setting -- first amplifier of series: Measure at the amplifier output and adjust a single input to give 3v rms. Do the same with the second input but with the first removed. With both inputs connected, the combined output should be about 4v rms.

Level setting -- second amplifier of series:

Measure at second amplifier output and adjust a single sine wave input to give 2.4v rms output. Do the same with the second input alone. Both inputs connected should give an output of about 3v rms. Disconnect both and connect the combined output from first amplifier and adjust to give an output of 3v rms. Reconnect the two single inputs and check the combined output of 4 signals. It should be about 4v rms. Check on an oscilloscope will show the signal to be about 20v peak to peak.

Observed distortion --

Distortion observed on an oscilloscope as peak clipping is relatively severe, about ½% when first noticeable. The distortion at which this amplifier is intended to operate is 0.1% or less of 2nd or 3rd or higher harmonic distortion and less than 0.15% of any intermodulation product. The onset of distortion is quite marked. Dropping the voltage level 1/8 below that at which peak clipping is first discernable will usually reduce the distortion from ½% to less than 0.1%.

COMPONENTS

A list of the drawings, sketches, and specifications comprising the CTOIS is given in Appendix IV. For convenience, these are segregated according to the following classifications:

GENERAL INSTALLATION

TOWED ELEMENTS

HANDLING ELEMENTS

ELECTRONIC ELEMENTS

Drawing numbers for the details of the 20,000-pound line pull capacity Linear Cable Engine designs are not included.

General information relative to the major items of equipment designed for the CTOIS may be found in Appendix V.

INSTALLATION DATA

Practically all data required for the mechanical installation is given on SED Drawing 50091. Information concerning weights, sizes, and center-of-gravity locations for specific items of equipment may be found in Appendix V. Comments concerning general arrangement and installation requirements are given below for consideration in the event the arrangement requires modification.

The basic consideration is to place the linear traction engine in a bight of the faired tow line such that the tensioned line bears against the vertical rollers situated on the side of the cable engine opposite the open face of the tracks. The angle between the centerline of the tracks and the centerline of the stern-sheave groove should be small (2 to 5 degrees) to prevent excessive load on the vertical entry roller and to avoid turning the fairing to a vertical position, as would occur if the bend becomes too sharp.

The bend in the cable may be much sharper on the low tension side without over-loading the vertical guide roller. The storage reel is positioned so that the angle varies from zero to fifteen degrees as the cable travels from the outboard to the inboard side of the storage reel. It should be noted that if the cable is allowed to bend away from the cable engine, the trailing edge of the fairing will bear against the outer vertical roller.

The fairing may tend to "flip" up when the cable is bent sharply as at the 15-degree angle. In this case, auxiliary guide rollers may need be installed to assure proper orientation. The proper arrangement of such rollers, if required, is best determined on site.

It will also be found that the fairing will turn under the cable in the unsupported length between the sheave and the cable engine. It will always turn counter-clockwise (looking aft) during payout as the trailing edge is situated to the starboard side of the cable as it exits from the cable engine. The trailing edge must then be picked up by the stern-sheave flange. This may not occur if the stern sheave flange becomes roughened and assistance may therefore be required. The extent of such assistance is best determined and provision therefore made on site.

On the other hand, during inhaul, the fairing must be forced to turn under in the counter-clockwise direction (looking aft) in order to enter the cable engine properly. Hence, if auxiliary means for alignment of the fairing with the sheave groove is made, it should provide also for forcing the fairing to turn counter-clockwise during inhaul.

A technique for coercing the fairing rotation may be of interest. If the axis of any vertical guide roller against which the fairing bears is inclined from the perpendicular to the cable-axis, so that the cable in passing must slide somewhat relative to the axis of the roller, the friction between the fairing clips and the roller will tend to turn the fairing in the direction of inclination of the roller axis as viewed in the direction of motion of the cable.

OPERATIONAL CONSIDERATIONS

The CTOIS, like any towed system utilizing faired cable, should always be launched and retrieved with way on the vessel, as this provides for proper streaming of the depressor and fairing. If the vessel comes dead in the water, the hydrodynamic resistance of the depressor to cable-torque is lost, and the cable can be expected to take a hundred turns or more if at full scope.

A speed of about three knots should prove comfortable for these operations. Minor adjustments in shaft R.P.M. may be made to acquire the most comfortable situation. In heavy seas, it may prove profitable to run with the sea at about wave speed to acquire a steady base for the operation, particularly when the depressor is near surface and hence, subject to large variations in force due to vertically imposed motions. If such a situation cannot be avoided, a drogue rigged to a large block and permitted to ride down the wire to the depressor will provide some relief by shallowing the wire angle due to the additional drag.

Care must be taken during retrieval not to allow the ships head to fall off in a high wind as this can result in an apparent set to port or starboard of the tow. The wire will then lead in on a cant relative to the sheave axis which, if sufficient, can result in the sheaves carrying the wire over its lip and into the guard. This is not as important during launch operations as the wire then enters the sheave on the proper track.

Once the tow is set, it is wise to rig a keeper on the wire aft of the stern sheave for extended cruising. This is not provided in the basic design, but can consist of a block of about a foot in diameter, rigged on a short chain bridle with the legs secured to the sides of the after-part of the sheave frame. A convenient method is to rig the block while at low speed and secure the legs of the bridle so that the block rides easily. The wire will then be cinched down at the higher towing speeds, at which shallower wire angles obtain and so secured against any possibility of leaving the sheave if bad weather is encountered. The length of the legs should be adjusted on the basis of observation so as not to bring the wire in contact with more than 15 to 20 degrees of the block-sheave at the higher speeds. The block should have a high crown to permit accommodation of the fairing between the cheeks. It will be found that with care minor adjustments of cable length can be made with the rig in place.

It will be found that the fairing will approach the sheave in an inverted position during inhaul and hence will require careful observation during that operation. Close coordination must be maintained with the winch operator so that the fairing may be properly oriented manually in the event it does not respond to the lifting effect provided by the sheave groove.

Here, again, retrieval at low ships speed is advised as the wire assumes a steeper angle and the fairing has the least tendency to turn under. A method for automatic righting of the fairing might be devised, but this writer knows of no convenient and acceptable scheme.

Also, during inhaul, a twist in the fairing will on occasion be encountered, and further down the line, a twist of opposite sense. This occurs due to the combining of the erratic nature of the wake of the vessel with the tendency of the fairing to hang "tail down", on the unsupported length of cable between the sheave and the water to "flop" a short length through 360 degrees. The lower twist tends to "corkscrew" down the cable until stopped by excessive friction between the clips and cable or until the cable angle becomes rather steep. When such a condition is encountered, it becomes necessary to manually orient the fairing on its entry to the stern sheave.

It will be found desirable to slow the line-speed during inhaul in traversing a cable connector over the stern sheave, through the traction unit, and onto the cable storage unit. Some difficulty may be encountered with the connector assembly in traversing through the level wind onto the cable storage reel when the level wind carriage is at the extreme port side. Provision has been made for manual operation of the level-wind drive to permit shallowing the entry angle if happenstance so requires. The drive may be returned to automatic as soon as normal winding is restored.

DISCUSSION

Some comments concerning the design are given below:

1. It appears desirable to move Station No. 6 from the location shown on Drawing 50089 to a point just above the depressor retrieval line to avoid the problem of orienting the module while attending the cable and retrieval lines during the launch operation. This may be easily accomplished by increasing the length of the - 10 cable assembly to 30 feet. If this is done, the amount of fairing type 800045-3 called for in Technical Specification 8004 should be increased to 70 feet and the length of the 15-foot section of cable specified in Technical Specification 8003 increased to 35 feet.

2. The "eyelet-type" fairing hanger on the upper end of each cable assembly is placed a short distance below the fairlead, and the holiday filled with a short piece of the fairing 800045-3, to allow progressive "cut-back" of the cable for reseating of the cable lock in the event of fatigue and wear at that point.

3. The fairing hanger at the bottom of each cable assembly is not an "eyelet-type" and secures the fairing by means of clamps that engage the

Dacron-cord reinforcement in the leading edge of the fairing. This permits accommodation of the tolerance build-up in the fairing lengths and provides for shortening to accommodate reseating the cable lock.

4. The fairlead concept employed in the reported design represents a considerable simplification in arranging to pass a rigid section of line over a sheave. The mechanical design is untested, however, and it may be found that modification of the depth of imbedment of the load-carrying steel disks in the poly-estane matrix will be required to attain adequate stability with sufficient flexibility.

5. The most compact device found for termination of the armour wires is one manufactured by MECCA Cable and Service. This device is guaranteed to sustain 80% of the rated cable load. It is inexpensive but does require care in installation.

6. A moderate reduction of the size of the cable connector assemblies could, with the fairlead devices, permit use of drum-type tractive winches. This objective is highly desirable as the full line load can be developed with such winches without the

great length necessary with the linear cable engines. The 20,000-pound-line pull engine would need be nearly 30 feet in length to develop a 45,000-pound pull. The present connectors could be handled on a machine with five-foot diameter drums.

7. The fairing guide rollers shown on the 20,000-pound-pull linear cable engine will require redesign. These were not adequately specified in Technical Specification 8003. The rollers should be moved as close to the track entrances as space permits. The inner vertical guide roller need not be spring-loaded. A 3½-inch wide opening should be provided to permit the fairing to pass while laying horizontal. The outer roller should be given some "spring" to permit passage of the connectors and fairleads. The cable should traverse near the inner face of the track. Two horizontal rollers should be provided. They should lay close to the vertical rollers and, preferably, on the inboard side. The span of the outboard rollers should extend well beyond the width of the inboard pair to assure positive engagement between the outboard pair and the cable. This is particularly necessary on the low tension side of the winch as the cable tends to "walk" on the roller, evidently due to the lay of the outer wires.

8. In the deck arrangement requires fleeting the cable from the open side of the cable engine, the spring loaded-roller arrangement shown on Western Gear Drawing 102368 should be retained, as it will then be necessary to turn the tail of the fairing upward for passage. If retained, they should be lengthened and, if possible, set closer to the horizontal rollers. The load carrying horizontal roller may be placed in a fixed frame and the opposite rollers spring-loaded to permit passage of the connector and fairlead.

9. Satisfactory performance of the cable and instrument housing combination is based on the assumption that the torsional characteristics of the cabling will not be more severe than those of the 0.782-inch diameter cable used to obtain a measure of those characteristics. This is plausible, as the slightly smaller diameter of the specified cable should result in slightly smaller values for the driving torque ($\bar{r} \tan \varphi$) and the "geometric" stiffness ($\bar{r}^2$), as the construction is expected to be similar. If greater values obtain for the purchased cable, the predicted performance may be retained by simply increasing the hydrodynamic stiffness of the housing so that the increased stiffness is in the same ratio to the existing

stiffness as the increased stiffness of the cable is to that reported herein.

10. The modifications to the depressor called out in Drawing 101514 were designed to provide for increase in the effectiveness of the horizontal stabilizer with no major change in the ballast. The zero-speed trim should be checked, however, and a small weight added as required, to bring the trim to about five degrees down by the nose.

CONCLUSION

A system has been designed that will permit sound velocity measurements to be taken simultaneously at six depths and to depths as great as 5,000 feet while underway at a speed of 7 knots. The design represents a state-of-the-art system in that it can be constructed using ordinary processes and materials. Although the design is slanted toward acquisition of sound-velocity data, any suitably housed instrumentation having output characteristics compatible with the electrical characteristics of the cabling can be utilized.

Several advancements in the state-of-the-art are believed to be represented by the system.

1. A "directed-flow" sound-velocity instrument container has been developed, which provides for virtual elimination of flow-induced error in sound-velocity measurement. The housing is adaptable with little additional engineering to use as a tethered probe that, with suitable winching equipment, can be deployed by a vessel while underway.

2. Techniques have been developed for prediction of the rotational and torsional characteristics of long wire ropes and for prediction of the destabilizing moments induced by curvature of cable fairing.

3. A technique for accommodation over sheaves and drums of short, rigid sections of flexible cables has been devised.

4. Many of the problems incident to the mechanical handling, spooling and winching of long faired cables have been identified and solutions determined.

It is not to be expected that the system will initially be without "bugs." Several problems have been left for the development of "on-site" solutions. Experience may well serve to identify numerous areas in which modification will be desirable. It is believed, however, that the basic elements of a potentially valuable system for research in the problems of undersea war have been provided in a system of mutually compatible techniques and components.

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APPENDIX I

ANALYSIS OF THE ROTATION IN A LONG ROPE

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ANALYSIS OF THE ROTATION IN A LONG ROPE

THEORY

Consider a short length, Δs, of the $i^{\underline{th}}$ wire in a wire rope strand. Let L be the torque applied to the rope by a terminal fitting, as shown on Figure 1. For equilibrium,

$$L = \sum_i \left[r_i t_i \sin \varphi_i + C_i \frac{\Delta\theta_i}{\Delta s} \right] .$$

where t_i is the tensile force in a wire and r_i, θ_i and φ_i are as illustrated on Figure 1 and Figure 2. C_i is the torsional constant for the wire, i.e., the product of the elastic modulus in shear and the polar second moment of area. From the geometry, we may write $t_i = \Delta T_i/\cos \varphi_i$; hence,

$$L = \sum_i \left[r_i \Delta T_i \tan \varphi_i + C_i \frac{\Delta\theta_i}{\Delta s} \right] .$$

Also, since the "lay" of the wire will change due to the rotation,

$$\tan \varphi_i = \tan \varphi_{i_o} + r_i \frac{\Delta\theta_i}{\Delta s} ;$$

which, on substitution yields

$$L = \sum_i \left[r_i \Delta T_i \tan \varphi_{i_o} + \left(r_i^2 \Delta T_i + C_i \right) \frac{\Delta\theta_i}{\Delta s} \right] .$$

Here φ_{i_o} is the initial value of φ_i for $\Delta\theta_i/\Delta s = 0$ and θ_i is the angle of rotation at the point in question.

Performing the summation, indicating averaged values for the various constants by an over struck bar, and allowing $\Delta s \to 0$, we obtain

$$L = \bar{r}T \tan \varphi + (\bar{r}^2 \; T + \bar{C}) \frac{d\bar{\theta}}{ds} \quad , \qquad [1]$$

where the subscript on φ has been dropped.

Hence, dropping the bar on θ,

$$\theta = \int_0^s \frac{L - \bar{r}T \tan \varphi}{\bar{r}^2 \; T + \bar{C}} \, ds \quad . \qquad [2]$$

<u>SPECIAL CASES</u>

I. <u>Vertically Oriented Rope Sustaining A Freely Suspended Weight.</u>

For this case, $L \equiv 0$.

Suppose the upper end fixed, $s = 0$ at the upper end, and $T\ (s = 0) = T_o$. If the weight per unit length is ω, $T = T_o - \omega s$, and Equation [2] may be written as follows:

$$\theta(s) = -\int_0^s \frac{\bar{r}\ (T_o - \omega s)\ \tan \varphi\ ds}{\bar{r}^2\ (T_o - \omega s) + \bar{C}} \quad .$$

With the substitutions

$$s' = \frac{\omega s}{T_o} \;,\; ds = \frac{T_o}{\omega}\, ds' \;,\; \text{and } C = 1 + \frac{\bar{C}}{\bar{r}^2 \; T_o} \quad ,$$

we obtain

$$\theta(s')' = -\frac{T_0 \tan \varphi}{\bar{r}\omega} \left[\int_0^{s'} \left(\frac{1}{C-s'} - \frac{s'}{C-s'} \right) ds' \right] ;$$

which yields,

$$\theta(s') = \frac{T_0 \tan \varphi}{\bar{r}\omega} \left[(1-C) \log_e \frac{C-s'}{C} - s' \right] .$$

If

$$\bar{C}/(\bar{r}^2 \; T_0) \to 0, \; C \to 1, \text{ and } \theta(s') \to -\frac{T_0 \tan \varphi}{\bar{r}\omega} s' ;$$

or

$$\theta(s) = -\frac{\tan \varphi}{\bar{r}} s .$$

Hence, a heavily loaded rope simply unwinds until the effective "lay" disappears, the total number of turns, α, at any point s being simply $- s/p$, p being the effective length of pitch.

II. ROPE TERMINATED TO AN ELASTIC FOUNDATION.

Suppose the constraint from the foundation to be expressible in the form

$$L = -k\gamma$$

where k is the torsional stiffness and γ the angle of rotation. Equation [2] then becomes

$$\theta(s) = \int_0^s \frac{-k\gamma - T\bar{r} \tan \varphi}{\bar{r}^2 \; T + \bar{C}} ds .$$

For purpose of illustration, let us suppose $T(s) = T_0$.

$$\theta(s) = \left(\frac{-k\gamma - T_0 \bar{r} \tan \varphi}{\bar{r}^2 \; T_0 + \bar{C}} \right) s .$$

But $\theta\ (s = \ell) = \gamma$, hence

$$\gamma = -\frac{T_0\ \bar{r}\ \tan\varphi}{\frac{\bar{r}^2\ T_0 + \bar{C}}{\ell} + k} \quad .$$

III. ROPE TERMINATED TO RIGID FOUNDATIONS.

In this case L is indeterminate. However, from Equation [1],

$$\frac{dL}{ds} = \bar{r}\ \tan\varphi\ \frac{dT}{ds} + \bar{r}^2\ \frac{dT}{ds}\ \frac{d\theta}{ds} + (\bar{r}^2\ T + \bar{C})\ \frac{d^2\theta}{ds^2} = 0 \quad .$$

Assuming the origin of the coordinate s at the lower end of the rope and $T = T_0 + \omega s$, we set the expression for the torque in the form

$$[G + S]\ \frac{d^2\theta}{dS^2} + \frac{d\theta}{dS} + \frac{\ell\ \bar{r}\ \tan\varphi}{\bar{r}^2} = 0 \quad , \qquad [3]$$

where

$$S = s/\ell, \quad \Delta T = \ell\omega \quad ,$$

and

$$G = \frac{1}{\Delta T}\left[T_0 + \frac{\bar{C}}{\bar{r}^2}\right] \quad .$$

Fortunately, the variables are separable in Equation [3], hence, double integration yields

$$\theta = C_1\ \log\ (G + S) - AS + C_2 \quad , \qquad [4]$$

where $A = \ell\ \bar{r}\ \tan\varphi/\bar{r}^2$ and C_1 and C_2 are the constants of integration. "Log" is of course $\log_e$.

Setting $\theta(S = 0) = 0$ yields $C_2 = - C_1 \log G$; and $\theta(S = 1) = 0$ yields $C_2 = - C_1 \log(G + 1) + A$. These are satisfied only if

$$C_1 = \frac{A}{\log\left(\frac{G + 1}{G}\right)} \quad .$$

Hence

$$\frac{\theta}{A} = \frac{\log\left(\frac{G + S}{G}\right)}{\log\left(\frac{G + 1}{G}\right)} - S \quad , \qquad [5]$$

which satisfies the conditions $\theta(S = 1) = \theta(S = 0) = 0$. From Equation [5],

$$\frac{1}{A}\,\frac{d\theta}{dS} = \frac{1}{(G + S)\,\log\left(\frac{G + 1}{G}\right)} - 1 \quad . \qquad [6]$$

The position of maximum rotation is found to be

$$\bar{S} = \frac{1}{\log\left(\frac{G + 1}{G}\right)} - G \quad . \qquad [7]$$

from Equation [6]. Substitution of $\bar{S}$ for S in Equation [5] gives the maximum value of θ:

$$\frac{\theta}{A}\max = \frac{\bar{\theta}}{A} = \frac{\log\left[\dfrac{\left(\frac{G + 1}{G}\right)^{G}}{\log\left(\frac{G + 1}{G}\right)^{G}}\right] - 1}{\log\left(\frac{G + 1}{G}\right)} \quad . \qquad [8]$$

Moreover:

$$\lim_{G \to 0} \bar{S} = 0$$

$$\lim_{G \to \infty} \bar{S} = 1/2,$$

$$\lim_{G \to 0} \bar{\theta} = A,$$

and

$$\lim_{G \to \infty} \bar{\theta} = 0.$$

Since $G \to \infty$ is equivalent to the condition $\Delta T = 0$, we conclude that no rotation occurs in a rope under uniform tension.

Finally, writing Equation [1] in the form

$$L = T_0 \bar{r} \tan\varphi + \bar{r} \tan\varphi \, \Delta T \, S + \frac{\bar{r}^2 \, \Delta T}{\ell} [G + S] \frac{d\theta}{dS} \quad , \quad [1.1]$$

substituting for $d\theta/dS$ from Equation [6], and clearing the resulting expression, it is found that L is independent of length, as expected, and has the value

$$\bar{r} \tan\varphi \, [T_0 + \Delta T \, \bar{S}] \quad . \quad [9]$$

IV. ARBITRARY ANGLE OF TERMINATION

If we put $\theta(S = 0) = \theta_0$ and $\theta(S = 1) = \theta_1$ in Equation [4], the constants become

$$C_1 = \frac{A + (\theta_1 - \theta_0)}{\log\left(\frac{G+1}{G}\right)}$$

and

$$C_2 = \theta_0 - [A + (\theta_1 - \theta_0)] \frac{\log G}{\log\left(\frac{G+1}{G}\right)} .$$

Hence

$$\theta = [A + (\theta_1 - \theta_0)] \frac{\log\left(\frac{G+S}{G}\right)}{\log\left(\frac{G+1}{G}\right)} - AS + \theta_0 , \qquad [10]$$

and

$$\frac{d\theta}{dS} = \frac{(\theta_1 - \theta_0) + A}{(G+S)\log\left(\frac{G+1}{G}\right)} - A . \qquad [11]$$

V. TORSIONAL STIFFNESS

We consider the case of a rope with ends rigidly fixed in torsion and without relative angular displacement. As before let the load be given by $T_0 + w\ell S$. Designate an upper section by subscript "u" and a lower, by subscript "ℓ". The excess of internal torque resulting from an arbitrary rotation, θ, at the junction of "u" and "ℓ" is

$$\Delta L = L_u - L_\ell$$

From Equation [1.1] (noting $T_{o_u} = T_{o_\ell} + \omega \ell_\ell$, $S_u = 0$, and $S_\ell = 1$); the value of ΔL is found to be

$$\bar{r}^a \;\omega\; G_u \left(\frac{d\theta}{dS_u}\right)_{S_u = 0} - \bar{r}^a \;\omega\; (G_\ell + 1)\left(\frac{d\theta}{dS_\ell}\right)_{S_\ell = 1} .$$

Taking the value for $d\theta/dS$ from Equation [11] with $\theta_{1_u} = \theta_{o_\ell} = 0$ and $\theta_{o_u} = \theta_{1_\ell} = \theta$, gives

$$\Delta L = \frac{\bar{r}^a \Delta T_u}{\ell_u} A_u \left[\frac{1}{\log\left(\frac{G_u + 1}{G_u}\right)} - G_u\right] - \frac{\bar{r}^a \Delta T_\ell}{\ell_\ell} A_\ell \left[\frac{1}{\log\left(\frac{G_\ell + 1}{G_\ell}\right)} - (G_\ell + 1)\right]$$

$$- \left[\frac{\bar{r}^a \Delta T_u}{\ell_u} \; \frac{1}{\log\left(\frac{G_u + 1}{G_u}\right)} + \frac{\bar{r}^a \Delta T_\ell}{\ell_\ell} \; \frac{1}{\log\left(\frac{G_\ell + 1}{G_\ell}\right)}\right]\theta \quad . \qquad [12]$$

The sum of the first two terms on the right is just the difference in torque for zero rotation. It is accordingly designated ΔL_o. The coefficient of θ is just $d(\Delta L)/d\theta$, and is accordingly designated ΔL_θ. Hence, the torsional difference may be written in the form

$$\Delta L = \Delta L_o + \Delta L_\theta \; \theta \quad , \qquad [13]$$

where ΔL_θ represents the rotational stiffness.

If it is assumed that $G >> 1$, the series expression for the logarithmic function of G may be used in Equation [12]. Whence, after much manipulation and utilizing the definitions

of "A" and G as they apply to the two portions of the rope, we obtain, retaining only terms of order G^{-1},

$$\Delta L_0 = \bar{r} \ \tan \varphi \cdot \frac{\Delta T}{2} \left[1 - \frac{2}{3} G^{-1} \ (1 - 2S) \right] \quad , \qquad [14.1]$$

and

$$\Delta L_\theta = \frac{\bar{r}^2}{l} \Delta T \frac{G}{S(1 - S)} \quad . \qquad [14.2]$$

Here G, ΔT, etc., apply to the entire length of rope.

If we set $\Delta L = 0$ and solve for θ, we evidently obtain an alternate expression for the angle of rotation of a rope with fixed ends. Applying the assumption $G \gg 1$ to Equation [5], it is found that all terms of the resulting expression correspond with all terms of the alternate expression except those in which G occurs with exponent values less than minus one.

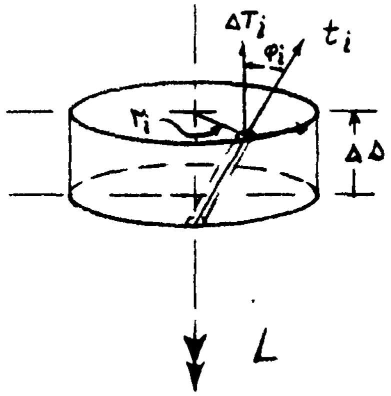

FIGURE 1

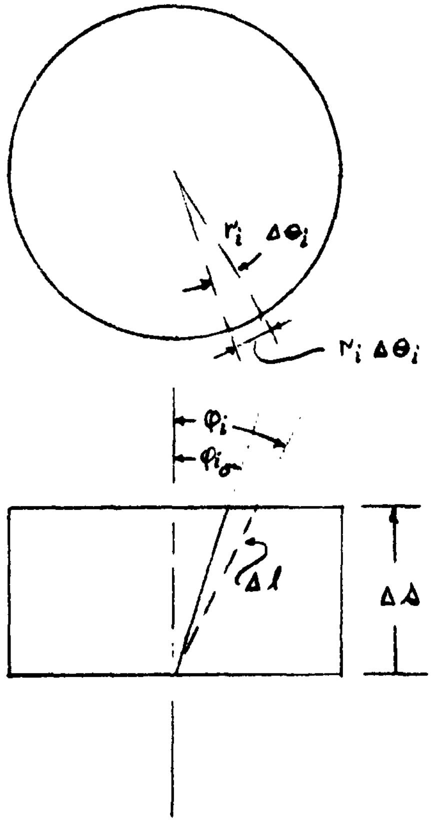

FIGURE 2

DEFINITION SKETCHES FOR DERIVATION

OF ROPE - ROTATION THEORY

APPENDIX II

EXPERIMENTAL DETERMINATION

OF THE CONSTANTS

APPENDIX II

EXPERIMENTAL DETERMINATION OF THE CONSTANTS

DESCRIPTION OF THE EXPERIMENT

The constants needed to permit computation of the rotation were deduced from measurements of the angle and torque on a short length of cable of construction and size nearly that of the selected cable. The characteristics of the cable used are given in Table 1.

The cable was mounted vertically with the upper end secured against rotation and the lower end secured to a freely hanging weight pan. A cardboard tube encasing the cable was affixed to the cable about a foot below the upper end and terminated to a large disk mounted at right angles thereto near the mid-length. A second tube was similarly secured to the cable about a foot above the lower end such that the disk attached thereto was located directly under that first mentioned. The disks were calibrated in degrees. The zero-twist position was determined from the relative position of the disks when the load was removed from the lower end of the cable.

Load was then applied to the lower end of the cable and allowed to oscillate. The successive peak amplitudes were observed and recorded. Torque was then applied to the weight and the average amount needed to null the rotation determined for two values of the applied tension.

The angle corresponding to zero torque, $\theta(L = 0)$, was taken as the average of the mid-points of the successive peak to peak amplitudes for each half cycle. Peak amplitude values of the first cycle were not used and of course the last value was discarded as its position within the dead-band can not be ascertained. Even numbers of data points were used in each case (six points, actually) since it can be shown that use of an odd number to determine the mid-point of the oscillation of a system with solid friction results in error of the order r/nk, where r is the frictional torque, k the spring constant for torsion, and n the number of data points. The mid-points thus deduced were averaged and subtracted from the average value of the reference angle for zero load to obtain $\theta(L = 0)$. The resulting data are given in Table 2, and plotted on Figure 1.

ANALYSIS OF THE DATA

In keeping with Equation [1] of Appendix I, it was assumed that the torque due to twist of a cable under uniform load varies linearly with θ. Hence, from Equation [1] of Appendix I,

$$\frac{\partial L}{\partial T} = \bar{r} \tan \varphi + \bar{r}^2 \frac{\theta}{\ell}$$

and so

$$(\partial L/\partial T)_{\theta = 0} = \bar{r} \tan \varphi$$

From the plot of L versus T shown in Figure 1.2, we find $\bar{r} \tan \varphi = 0.0169$ inches.

Moreover $\frac{\partial}{\partial T} \left|\frac{\partial L}{\partial \theta}\right| = \frac{\bar{r}^2}{\ell}$. From Figure 1,

$\ell \left|\frac{\partial L}{\partial \theta}\right|_{T = 3120 \text{ lb}} = 1159 \text{ lb-in.}^2$, and

$\ell \left|\frac{\partial L}{\partial \theta}\right|_{T = 2200 \text{ lb}} = 1062 \text{ lb-in.}^2$. Hence

$$\bar{r}^2 \cong \frac{(1159 - 1062) \text{ lb-in.}^2}{840 \text{ lbs}} = 0.1152 \text{ in.}^2 \quad .$$

Thus,

$\bar{r} = 0.34$ inches and $\varphi = 2.84$ degrees. Both are plausible.

The value of $\bar{C}$ may be found by either of two methods. We have $\ell \left|\frac{\partial L}{\partial \theta}\right| = \bar{r}^2 T + \bar{C}$. This may be solved directly for $\bar{C}$ as $\bar{r}^2$ is known. On the other hand, we may eliminate $\bar{r}^2$ if measures of $\ell(\partial L/\partial \theta)$ are available for two values of T. This gives the result

$$\bar{C} = \frac{\ell\left[\left|\frac{\partial L}{\partial \theta}\right|_1 - \frac{T_1}{T_2}\left|\frac{\partial L}{\partial \theta}\right|_2\right]}{1 - T_1/T_2} \quad .$$

Using the values for $\ell(\partial L/\partial \theta)$ given above, we find three values of $\bar{C}$: 799 lb-in.^{2}, 799 lb-in.^{2}, and 802 lb-in.^{2}. The first two values were obtained by the method first mentioned and the last by means of the equation immediately above. The actual value was taken as the average of the three figures, 800 lb-in.^{2}.

TABLE 1

CHARACTERISTICS OF THE TEST CABLE

Overall Diameter	=	0.782 inches
Weight Foot	≅	0.8 lbs/ft
Test Length	=	80.5 inches
O.D. Outer Wires	=	0.083 inches
O.D. Inner Wires	=	0.083 inches
Pitch Angle, outerwires	≅	20° Left lay
Pitch Angle, innerwires	≅	20° Right lay
Core Diameter	≅	0.450 inches
Rated braking load	≃	45,000 lb

Material - Extra Improved Plow Steel

Galvanized

TABLE 2

RESULTS

TENSION Pounds	TORQUE/lb-in. $\theta = 0$	ANGLE/DEGREES Torque = 0
2280	38.5	2.92
3120	52.9	3.67

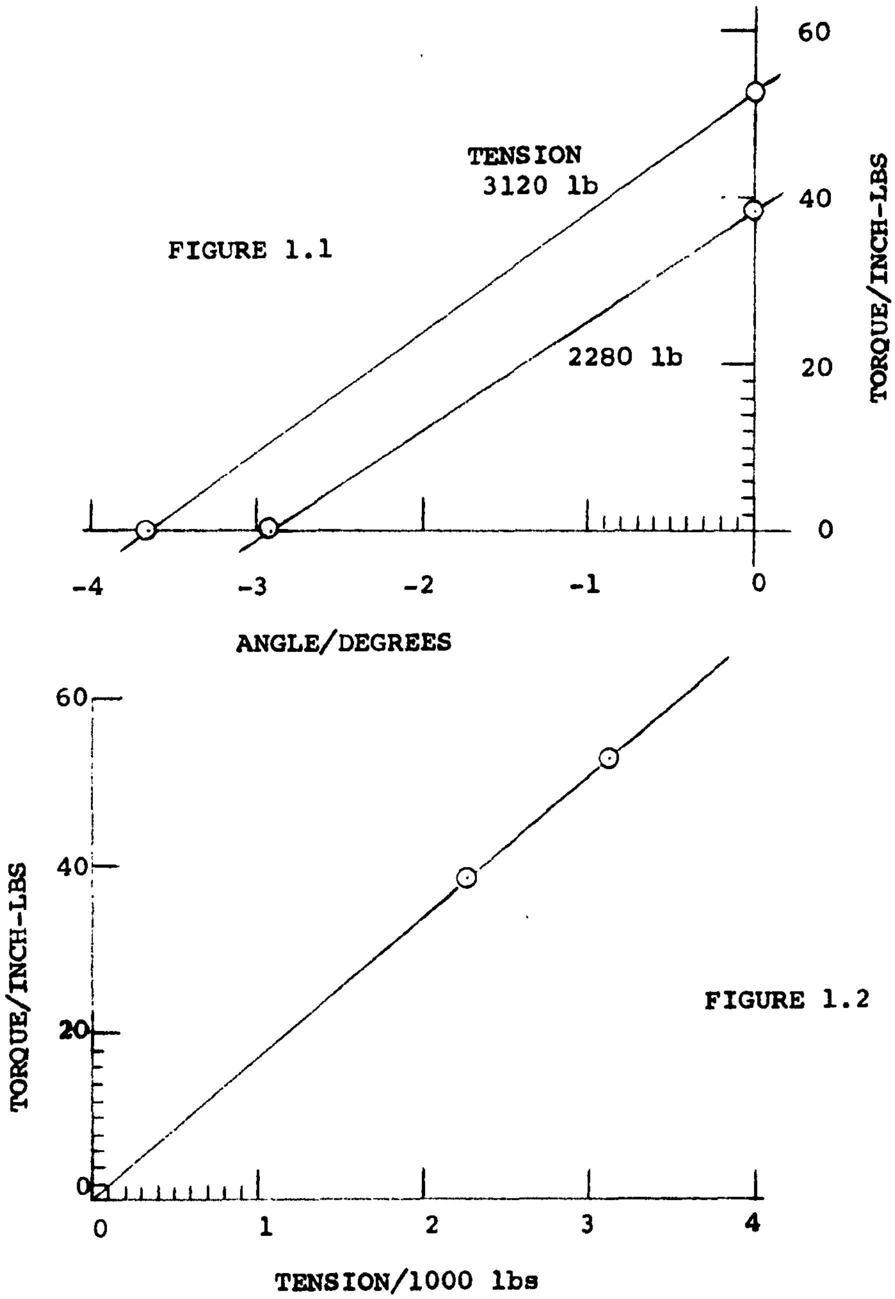

FIGURE 1 - PLOTS OF TEST DATA

APPENDIX III

EFFECT OF CABLE ROTATION

ON MODULE ORIENTATION

APPENDIX III

EFFECT OF CABLE

ROTATION ON

MODULE ORIENTATION

Given an instrument module trailing a segment of cable. The initial configuration is shown on Figure 1. The coordinate Z_1 theron is oriented vertically and the positive direction is down. Coordinates (X_1, Y_1, Z_1) are fixed; coordinates (X, Y, Z) move with the module. Coordinates Y and Y_1 are normal to the planes X, Z and X_1, Y_1 respectively. The angle φ is the inclination of the cable to the horizontal and α_o, the angle attack of the module.

If the cable rotates through an angle θ (taken as positive for clockwise rotation when viewed in the direction of positive S), the plane X, Y is rotated through the same angle (see Figure 2). The Y axis is thereby inclined through the angle ω with respect to the plane X_1, Y_1, where

$$\sin \omega = \cos \varphi \sin \theta \qquad . \qquad [1]$$

The X axis then forms the angle β with X_1, where

$$\cos \beta = \cos \varphi \cos \zeta + \sin \varphi \sin \zeta \cos \theta \qquad ; \qquad [2]$$

Y forms the angle γ with X_1, where

$$\cos \gamma = \sin \varphi \sin \theta \qquad ; \qquad [3]$$

and Z forms the angle ϵ with X_1, where

$$\cos \epsilon = \cos \varphi \sin \zeta - \sin \varphi \cos \zeta \cos \theta \, . \qquad [4]$$

The angle of attack of the module is given by

$$\tan \alpha = \cos \epsilon / \cos \beta \qquad [5]$$

and the angle of sideslip by

$$\tan \psi = \cos \gamma / \cos \beta \, . \qquad [6]$$

The side-force on the module is given by

$$Y = Y_\psi \, \psi - W \sin \omega \qquad [7]$$

and the moment about the cable by

$$L_M = W \sin \omega \, \ell_W \sin \zeta - Y_\psi \, \psi \, \ell_y \sin \zeta \quad ; \qquad [8]$$

where Y_ψ is the rate of change of Y relative to ψ, W, the net weight in water, and ℓ_y and ℓ_w are the respective distances to the center of action of Y and W measured along X from the intersection of S with X.

The component of force on the module normal to the X, Y plane is given by (for small α)

$$Z = Z_\alpha \, \alpha + W \cos \lambda \quad , \qquad [9]$$

where

$$\cos \lambda = \sin \zeta \sin \varphi + \cos \zeta \cos \varphi \cos \theta \, ,$$

and the moment about the intersection of S with X by

$$M_y = (Z_\alpha \, \alpha \, \ell_Z + W \cos \lambda \, \ell_W) \quad , \qquad [10]$$

where Z_α and ℓ_Z are the equivalents of Y_ψ and ℓ_y .

But cos ϵ, cos β and cos λ vary as θ^2. Hence, to the first order of smallness in θ,

$$\zeta = \varphi - \frac{W\ell w}{Z_\alpha \ell_Z} = \varphi - \alpha_o \quad , \qquad [11]$$

as $M_y = 0$. On this condition,

$$L_m = (W\ell_w \cos\varphi - Y_\psi\ \ell_y \frac{\sin\varphi}{\cos\alpha_o})\ \sin(\varphi - \alpha_o) \times \theta \ . \qquad [12]$$

Otherwise ζ must satisfy the relation

$$- \frac{\cos\epsilon}{\cos\beta\ \cos\lambda} = \alpha_o \quad . \qquad [13]$$

APPENDIX III

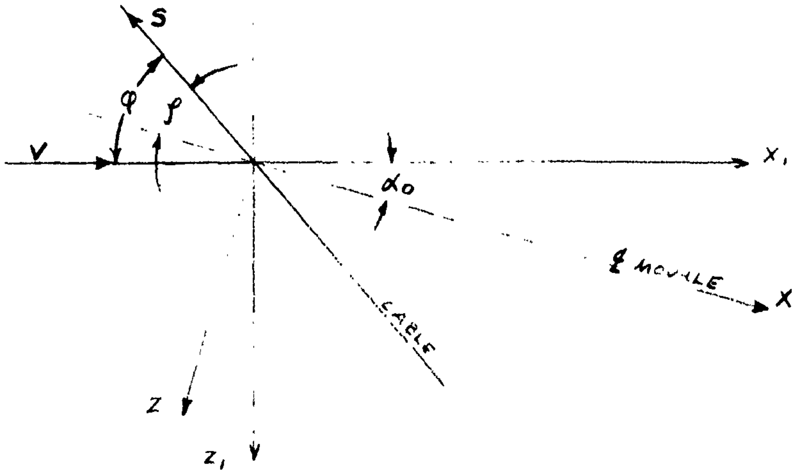

FIGURE 1 - DEFINITION SKETCH SHOWING CONFIGURATION
FOR NO ROTATION

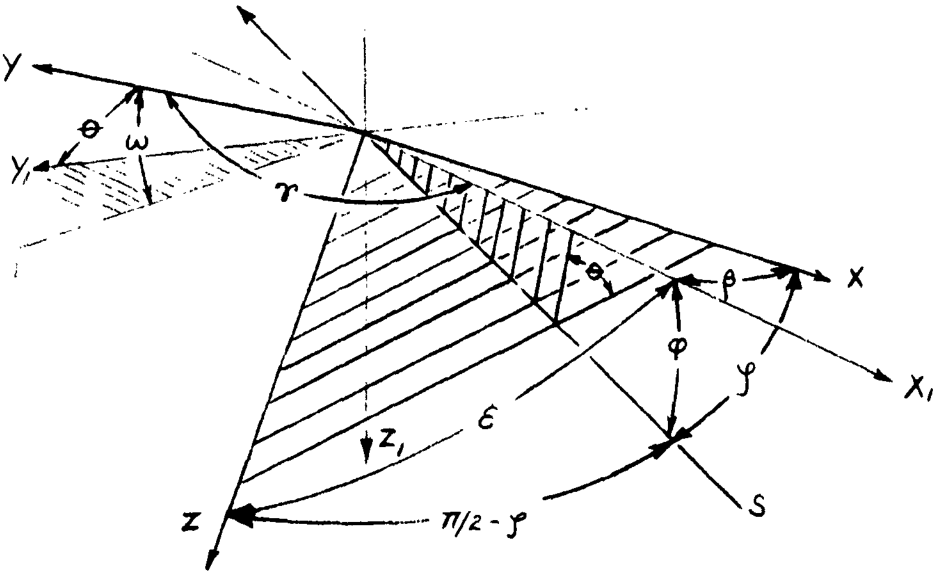

FIGURE 2 - DEFINITION SKETCH SHOWING CONFIGURATION
AFTER A ROTATION θ

APPENDIX IV

DRAWINGS, SKETCHES, AND SPECIFICATIONS

APPENDIX IV

DRAWINGS, SKETCHES, AND SPECIFICATIONS*

I. GENERAL INSTALLATION

DRAWING NUMBER	TITLE
50091	Deck Arrangement - Towed Array Handling Equipment - Josiah W. Gibbs

II. TOWED ELEMENTS

DRAWING NUMBER	TITLE
50089	Assy, Towed Elements
101116	Fairing Splice
101407	Nose Casting
101443	Screw, Swivel Lock
101451	Swivel
101501	Connector Module Details
101502	Stop, Swivel
101503	Instrument Module Assy.
101504	Instrument Housing
101505	Bulkheads
101506	Wing
101507	Spring, Cocking
101508	Retainer, Cable Spring
101509	Spring, Cable
101510	Nut, Lock
101511	Shell, Connector
101512	Clip, Tag Line
101513	Details, Tow Staff
101514	Depressor, Modification
101515	Clip Installation
101516	Stop

* Unless otherwise noted, all drawings, specifications, etc., are by PneumoDynamics Corporation.

DRAWING NUMBER	TITLE
101517	Clamp & Stop Installation
101604	Details, Instrument Mod
101609	Connector, Tee
101610	Cable Assy, Interconnecting
101611	Fairlead, Molded
101620	Nameplate - CTOIS
101647	Screw, Lock, Swivel
101649	Clip, Filter Mt'g - CTOIS
101648	Swivel - DTA
101650	Stop - Swivel - DTA
800043	Cable
800045	Fairing

SPECIFICATION	TITLE
8004	Specification - Trailing - Type Cable Fairing for Cable-Towed Oceanographic Instrumentation System
8005	Specification - Hydrographic Cable for Cable-Towed Oceanographic Instrumentation System

III. HANDLING ELEMENTS

A. LINEAR CABLE ENGINE

DRAWING NUMBER	TITLE
102368*	Linear Cable Engine (Assembly Drawing)

Instruction Book and Spare Parts List

Manufacturers Book No.	Instruction Book -
M.I.B. - 173	Linear Cable Engine

SPECIFICATION	TITLE
8003	Linear Cable Engine Specification for Deep-Towed Array Project

*Western Gear Corporation, Heavy Equipment Division, Everett, Washington

B. STORAGE UNIT

DRAWING NUMBER	TITLE
50085	Storage Unit Assembly, DTA
50090	Drive Installation, Storage Unit, DTA
101404	Reel, Storage Unit
101537	Boss Assembly, Fleeter Support Tube
101538	Gear Box, Spec. Control, DTA
101539	Jackshaft Details
101540	Details, Fleeter Drive
101541	Sprocket, Drive Assembly, DTA
101542	Details, Fleeter Assembly, DTA
101543	Fleeter Carriage Details, DTA
101544	Fleeter Carriage Frame Weldment, DTA
101545	Gear Box Weldments, DTA
101546	Gear Box Details, DTA
101547	Shaft Assemblies, Fleeter Drive, DTA
101548	Gear Box Assembly, DTA
101549	Main Frame Storage Unit
101550	Electric Schematic
101551	Hydraulic Schematic & Pump Unit Specification
101552	Fleeter Carriage Assembly & Installation, DTA
101553	Shaft Storage Reel, DTA
101554	Chain Guard, Main Drive, DTA
101555	Chain Guard, Fleeter Drive, DTA

SPECIFICATION	TITLE
8005	Storage Unit Procurement Specification For Cable-Towed Instrumentation System

C. STERN SHEAVE

DRAWING NUMBER	TITLE
101446	Sheave Assembly, Stern
101447	Sheave, Stern
101448	Pedestal, Stern Sheave
101449	Rope Guide, Stern Sheave
101450	Shaft - Stern Sheave

IV. ELECTRONIC ELEMENTS

Amplifier

DRAWING NUMBER	TITLE
SK 1468-2	Amplifier Schematic
SK 1470	Parts Layout, PC Board
SK 1471	Parts Layout, Transistor Mounting Plate
SK 1509	Wiring Connections, Transistor Mounting Plate
SK 1511	Emitters Follower Assembly
SK 1510	Emitter Follow Installation
SK 1469	Printed Circuit Layout
SK 1466	Mounting Plate, Transistor
SK 1505	Terminal Board
SK 1472	Input Transformer Data
SK 1473	Output Transformer Data
SK 1487	Bushing, 3/8" Lucite
SK 1489	Bushing, 7/8", Aluminum
SK 1490	Bushing, 1", Aluminum
SK 1512	Final Assembly
SK 1513	Level Adjustment Location
SK 1508	Parts List, DTA Amplifier

Power Supply

SK 1474	Power Supply Schematic
SK 1475	Parts Layout, PC Board
SK 1476	Parts Layout, Transistor Mounting Plate
SK 1477	Parts Layout, Rectifier Mounting Plate
SK 1514	Wiring Connections to PC Board
SK 1516	Transformer Wiring Connections
SK 1481	Printed Circuit Layout

Power Supply (continued)

DRAWING NUMBER	TITLE
SK 1467	Mounting Plate, Transistor
SK 1465	Mounting Plate, Rectifier
SK 1506	Terminal Board
SK 1486	Bushing, 5/16", Lucite
SK 1488	Bushing, 1/2", Lucite
SK 1489	Bushing, 7/8", Aluminum
SK 1490	Bushing, 1", Aluminum
SK 1517	Final Assembly, DTA Power Supply
SK 1518	Parts List, DTA Power Supply

Module

SK 1504	Wiring Diagram, DTA Module
SK 1515	Electrical Connector Location, DTA Module

APPENDIX V

COMPONENT DATA SHEETS

APPENDIX V

COMPONENT DATA SHEETS

Data sheets for the major pieces of handling equipment required for the CTOIS, certain electronic components are given herein.

I. Linear Cable Engine

Design: Western Gear Corporation, Everett Washington

Specification: PneumoDynamics Corporation, SED #8003

Description

Dimensions:

Height: 6' 7"

Width: 5' 8½"

Length: 13' 2" (exclusive of guide-roller assemblies)
Approximately 18' 3" overall

Weight: 16,300 lbs

Power required: 440 AC, 60 cycle, 3 phase

Performance:

Inhaul - 20,000 lbs line pull @ 50 f.p.m. @ 100 psig air pressure.

Backhaul - 1,000 lbs, line pull @ 100 f.p.m.

Maximum Speed - 100 f.p.m.

NOTE: See manufacturers instruction book M.I.B.-173 for more detailed information.

II. Cable Storage Unit

Design: Systems Engineering Division, PneumoDynamics Corp.

Specification: PneumoDynamics Corporation, SED #MS110-2-1

Identification: Drawing Number 50085

Description

Storage Reel

Volume: 326 cu ft

Diameter: 4' -0"

Width: 7' -0"

Flange Diameter: 8' -8"

Inertia: Full reel 8,400 lb-ft sec^2

Line Tension as Function of Line on Reel

First Layer: 1,000 lbs

Full Reel: 500 lbs

Line Speed, Inhaul, Payout: 0-100 ft/min

Overall Dimensions:

Length: 15' -8"

Breadth: 14' -0"

Height: 10' -2"

Weight (Less cable): 19,400 lbs

Center of Weight Location (Less cable)

Length: 3'-2" aft of reel axis

Breadth: 1' -11" starboard of centerline of reel

Height: 3' -4"

Weight (including 9,000 ft of 3/4-inch faired cable)

34,400 lbs

Center of Weight Location (with cable):

Length: 1' -10" aft of reel axis

Breadth: 1' -1" starboard of centerline of reel

Height: 4' 5"

Holding Brake

Type: Shoe, spring applied, hydraulic released

Static Torque: Design for 1,000 lbs line tension

Level Wind

Type: Chain and sprocket with traversing carriage

Capacity: 3/4" cable with TF-84 fairing also accommodates 4" diameter modules or cable connectors

Controls

Electric Motor: Start-stop switch for electric-hydraulic drive

Holding Brake: 440v/60 cy solenoid operated hydraulic valve

Auxiliary: Gear box for manual or automatic fleeting

Power Train

Prime Mover

Type: Electric-Hydraulic pumping unit, constant torque

Weight: 150 lbs + 450 lbs oil = 600 lbs

Electric Motor: 5 H.P., 440 AC, 60 cycle, 3 Phase, Marine Duty

Drive System

Type: Mechanical

Speed Reduction: 264/1 Gear box

Couplings: Geared flexible coupling

Dynamic Braking

Type: Hydraulic pump-motor

Power Absorption: 3 H.P.

Torque Available at Drum: 2,000 ft lbs

III. Stern Sheave

Design: Systems Engineering Division, PneumoDynamics Corp.

Specification: PneumoDynamics Corporation, SED #MS110-2,3

Identification: Drawing Number 101446

Description

Overall Dimensions:

Length: 6' -6"

Breadth: 3' -0"

Height: 7' -7"

Sheave: 6' -0" flange diameter x 5' -0" tread diameter x 1' -1" wide

Weight

Sheave: 2,480 lbs

Frame: 3,000 lbs

Bearings & Shaft: 262 lbs

Rope Guide: 204 lbs

TOTAL: 5, 946 lbs

Location of Center of Weight

Height: 2' -7"

Width: Centerline Sheave

Breadth: 5" fwd of sheave axis

Capacity

Operating Cable Tension 15,000 lbs

Maximum Cable Tension 45,000 lbs

IV. Instrument Module

Design: PneumoDynamics Corporation

Specification: PneumoDynamics Corporation, SED #MS110-1.2

Description

Length: 20-3/4"

Width: 20-3/4"

Height: 8-1/4"

Diameter: 4-1/2"

Weight: 26 lbs

Displacement: 8 lbs

Center of Gravity Location: 11.93 in. - aft of nose

Center of Buoyancy Location: 12.59 in. - aft of nose

Design Depth (Pressure Case): 8,000 ft

Minimum Towing Speed: 2 knots

Angle of attack @ 3 knots: 7 degrees (approx.)

Angle of attack @ 7¼ knots: less than 1 degree

Instrumentation

A.C.F. Model TR-4 Sound Velocimeter

Borg Warner Corporation Model 8150 "VIBROTRON" Pressure Transducer